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THE INFLUENCE OF FLUID FLOW THROUGH THE CLEARANCE SPACE ON THE MOMENT OF RESISTANCE TO ROTATION OF A DISC

Presenting General Results Of Experimental Investigations Of The Moment Of Resistance To Rotation Of Discs With Fluid Flow Between The Disc And The Casing

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Turbomachinery disc friction loss calculations usually make use of empirical equations derived from tests using discs rotating in closed casings. The wheel space of a conventional turbomachine passes leakage flows, in amounts depending upon the clearances and flow direction involved, which may substantially alter the moment of resistance to rotation of the disc as compared to the case with no flow.

The present work has considered the effect of through flow in the wheel space on the moment of resistance of smooth thin discs, when the flow is introduced at the center of the rotating disc, with no restricting clearance, and flows towards the rim. Of greatest practical interest is the case of turbulent flow in the wheel space, which, for a disc rotating in a closed casing, is established primarily for Reynolds numbers: (Ref. 1)

$$R_e = \frac{\omega R^2}{\nu} \geqslant 2.5 \times 10^5 \qquad (1)$$

Where: ω is the angular velocity of the disc, Rad/sec

R is the radius at the disc rim, meters

ν is the kinematic viscosity, M^2/sec

The characteristics of the regime with flow are not sufficiently defined by a Reynolds number based only on disc velocity, since the regime also depends upon the radial velocity of the stream in the wheel space.

In this case, it is necessary to consider the significance of the combination of the Reynolds number and the magnitude K, which appears as a ratio of the rim velocity of the disc to the mean radial velocity of the fluid through the gap at the rim.

$$K = \frac{\omega R}{V_{rm}} = 2\pi R^2 S \frac{\omega}{Q} \qquad (2)$$

Here: S is the width of the gap between the outside of the disc and the protruding surface of the casing, meters

Q is the volume flow of the fluid through the gap, M^3/sec

With decreasing K, the value of the Reynolds number at which the flow becomes turbulent is decreased, and it is possible to have turbulent flow in the wheel space with the disc stationary. (Flow in a plane radial diffuser).

Qualitatively, the effect of flow through the wheel space can be judged by comparing the tangential velocity profiles of the stream across

the gap for a disc rotating in a closed casing and for a disc immersed in flowing fluid. In a closed casing, the whole volume of fluid in the wheel space in the turbulent regime revolves with a mean angular velocity slightly less than one-half that of the disc. For this case, with $S/R = 0.005$ and $Re = 1.7 \times 10^6$, the experimentally determined distributions of tangential velocity for different gap widths and disc radii are shown by Figure 1. (Ref. 2).

Plotted along the ordinate are values of the ratio of the true tangential velocity of the stream, v_φ, to the velocity of the disc rim, $R\omega$.

With flow through the wheel space, the tangential velocity profiles of the stream in the gap are fundamentally different. (Fig. 2 and Ref. 2). First of all, the no-flow case strongly indicates a main stream with tangential velocity almost constant across the width of the gap at any radius, as seen in Figure 1. The restriction to the stream in the gap according to Figure 2 becomes less, which increases the velocity of the stream in the gap relative to the disc.

From Figure 2 it follows that even for moderate flow rates ($K=9$), the mean circumferential velocity of the stream in the wheel space does not exceed 0.06 times the disc rim velocity for r/R of 0.8. For very large flows, the main stream in the wheel space is practically unrestricted.

Since the friction moment of the disc depends to a certain extent on the difference in velocity between the disc and the main stream, the moment

of resistance to rotation of the disc can be higher with the presence of flow than for a disc rotating in a closed casing. The investigation of the dependency of the moment of resistance to rotation of the disc on the flow rate through the wheel space, gap width, and Reynolds number was carried out using a special experimental setup.

To determine the influence of the physical properties of the fluid on the moment of disc friction, tests were conducted with discs immersed in water and in air. The latter simultaneously extended the limit of Reynolds number in the experiments conducted.

Tests in water were conducted on a setup with a disc 155 mm in diameter which rotated at from 2000 to 5000 rpm, with gap widths from 1 to 40 mm, and water flow rates from 0.1 to 3.0 liters per second (see Fig. 3). In order to assure uninterrupted flow around the disc, the gap between the disc rim and the casing edge was maintained at a value of 0.5 mm. The flow was monitored with the aid of a 1 mm diameter opening at the center of the disc. (Part 6, Fig. 3). With gap width between the disc rim and the casing edge exceeding the upper limit given above, at small flows and large rpm, water flow from the opening is observed to cease, and, at the same time, other conditions remaining constant, the indicated friction moment varies erratically. The lack of a stable value of the friction moment in this case can be explained by the discontinuity of the stream in the wheel space.

Experiments in air were conducted with discs of two different diameters, (155 and 250 mm) rotating at speeds from 2500 to 7000 rpm and with gap widths

from 3 to 20 mm. Air flow rate was measured using a standard orifice and ranged from 0.03 to 0.12 (standard) M^3/sec. The scheme for introducing the air to the disc is shown by Figure 4. A shielding ring is placed to prevent the discharge flow from the gap from affecting conditions on the upper part of the disc.

The unique character of this setup appears in the introduction of the electric current to the open-type motor through mercury-filled plexiglas rings (Parts 9 and 14, Fig. 3), and in the measurement of the friction force moment by means of the angle of twist of the torsion dynamometer spring (string) Part 17, which properly receives the weight of the movable parts of the arrangement and decreases the friction moment in the radial thrust bearing. The character of the fluid motion in the gap is essentially dependent upon the restriction of the gap to the stream and the radial component of velocity in the vicinity of the disc surface, which is determined by the relationship between the mass (centrifugal) forces and the viscosity and inertia forces.

Friction in the vicinity of the disc depends upon the velocity of the stream in the gap relative to the rotating disc. In this relative motion, the centrifugal forces appear as mass. Therefore, Galeele's criterion acquires meaning with the substitution of centrifugal force for gravitational mass force. The gravitational mass force, centrifugal force in our problem, is proportional to the first power of the lineal dimension (radius).

Besides the well-known geometric, kinematic, and dynamic similarity

criteria for rotating disc problems, (s/R, K, Re), the dynamic force similarity criterion of Galileo enters into calculations for the moment of resistance to rotation of a disc in the presence of flow. This criterion is represented by the ratio of the square of the Reynolds number to the Froude number.

$$Ga = \frac{Re^2}{Fr} = \frac{g s^3}{\nu^2} \qquad (3)$$

where $g = 9.81$ M/sec^2

With flow through the gap, the increased moment of resistance of the rotating disc as compared to the no-flow value depends only on the Ga number. Therefore, it is convenient to consider a friction force moment coefficient with flow, made up of the sum of the friction coefficient with no-flow and an increment, depending on the amount of flow.

$$C_f = C_{fo} + \Delta C_f \qquad (4)$$

The coefficient of the friction force moment is defined by the formula:

$$C_f = \frac{M}{\rho \omega^2 R^5} \qquad (5)$$

where M is the moment of resistance to rotation of one side of the disc, Kg-M.

ρ is the density of the flowing fluid, Kg-sec^2/M^4.

The coefficient of the friction force moment, C_f, increases with an increase in flow rate, but at small flows the rate of increase is considerably greater than for large flows.

With increased Reynolds number due to an increase in rpm, the increase in ΔC_f becomes less. It should be noted that the increase in ΔC_f becomes greater with larger disc radii, and smaller kinematic viscosity. (Influence of Cs).

The last is particularly illustrated in the experimental results with water and air, with identical disc diameters, gap width, and rpm. In the case of the tests with water, the maximum increase in the coefficient (ΔC_f) exceeds the coefficient of the friction force moment for the no-flow case, C_{fo}, by four times, while in air, ΔC_f is greater than C_{fo} by only six times, although the flow through the setup was forty times as great.

The absolute value of the increase in the coefficient of the friction force moment due to so large a leakage through the wheel space strongly shows the necessity of taking wheel space flow into account in turbomachinery disc friction loss calculations. With decreased gap width, a decrease in the magnitude of C_f is observed only until a definite value of S/R is reached, after which it again rises, since the interaction of the disc boundary layer with that of the casing wall begins. For a disc 155 mm in diameter, with $Re = 1.5 \times 10^6$, the maximum value of the coefficient C_f is seen at relative gap width $S/R = 0.026$, which coincides with the minimum value found experimentally for a disc rotating in a closed casing. One may conclude from this that the optimum val e of the gap width from the point of view of minimizing the friction moment ? the disc for a given disc radius can be defined by the formula:

$$\frac{S}{R}\,\text{opt} = \frac{3}{\sqrt[3]{Re}} \qquad (6)$$

Graphic results summarizing more than 640 tests conducted in water and air, for $K = 0.6$ to 8000, $S/R = 0.013$ to 0.52, and $Re = 10^5$ to 3×10^6 are presented by Figure 5.

As a result of the present work the following formula is derived for defining the increase in the friction force moment coefficient:

$$\Delta C_f = 0.42 \times 10^{-3} \frac{\left(\frac{S}{R}\right)^{0.75} Ga^{0.3}}{K^{0.8}} \qquad (7)$$

The mean deviation of the experimental points in Figure 5 from a straight line described by this equation is not more than 10%. The fact that the results of tests with fluids of different physical properties (water and air), conducted with two different diameter discs, at various gap widths and rpm, lie on one straight line, indicates that the accepted criteria of similarity satisfactorily reflect the physical substance of the phenomena of our problem.

The following formula is recommended for defining the coefficient of the friction force moment on one side of a disc with wheel space flow:

$$C_f = \left[\frac{0.151}{\left(\frac{S}{R}\right)^2 Re^{1.2}} + \frac{1.02 + \frac{S}{R}}{\left[72 + 12\,\frac{S}{R}\right] Re^{.182}}\right] + \Delta C_f \qquad (8)$$

The bracketed expression by itself is that presented by Pantell for discs rotating in closed casings. (Ref. 3).

For small ratios of S/R, insignificantly greater than $(S/R)_{opt}$, it is convenient to use the greatly simplified formula:

$$C_f = \frac{0.0187}{\sqrt[5]{Re}} + \Delta C_f \qquad (9)$$

The results of our experiments can be used in cases where the fluid is introduced into the wheel space through round pipes or a ringed slot with relative radius of such a magnitude that it is considerably larger than in our experiment $\frac{r_m}{R} = 0.3$, since in formula (5) for the moment of resistance to rotation, the radius enters in the 5th power. If, for example, the fluid is introduced through a narrow ringed slot with a mean radius equal to one-half that of the disc, we can expect a decrease in the moment of friction force on the order of 3%.

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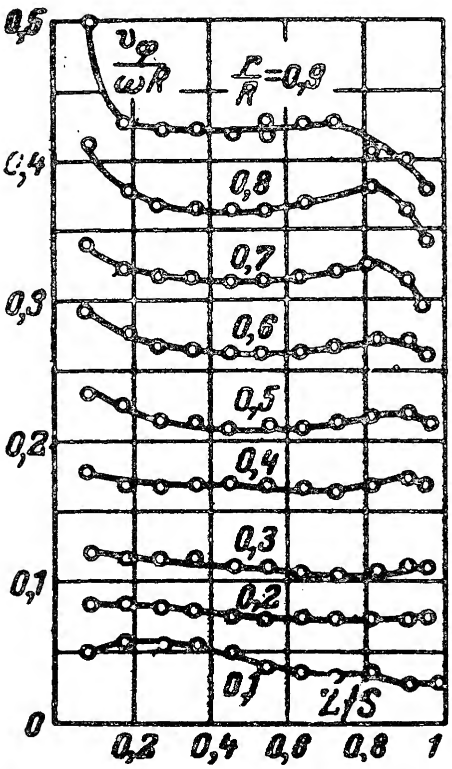

Fig. 1 - tangential velocity profile of the stream in the gap between the rotating disc and the walls of the casing for the no-flow case.

$Re = 1.7 \times 10^6$; $S/R = 0.055$

r = Present value of radius

z = Distance from the point to the disc

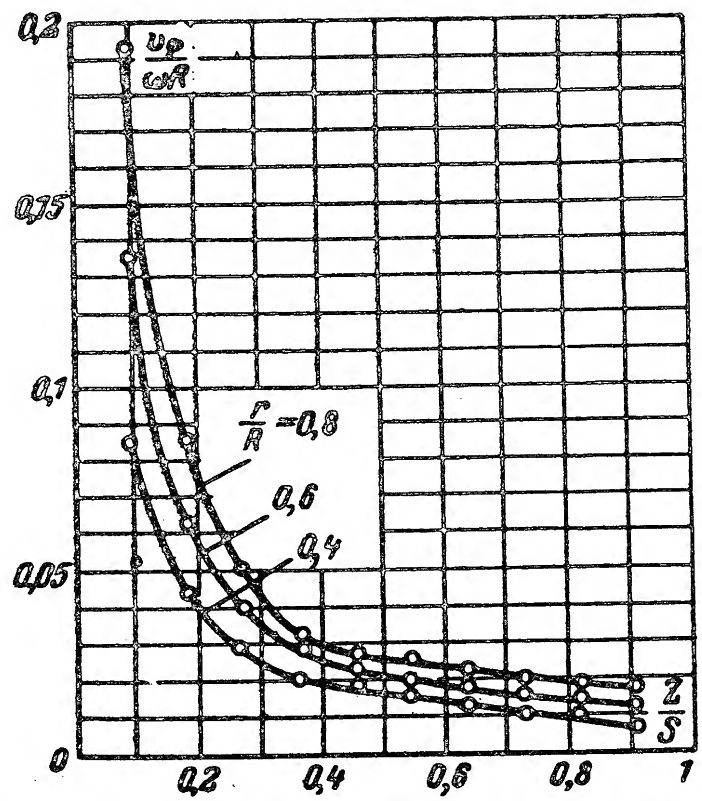

Fig. 2 - tangential velocity profiles in the gap between the disc and the casing wall with flow.

$Re = 1.1 \times 10^6$; $S/R = 0.055$, $K = 9.0$

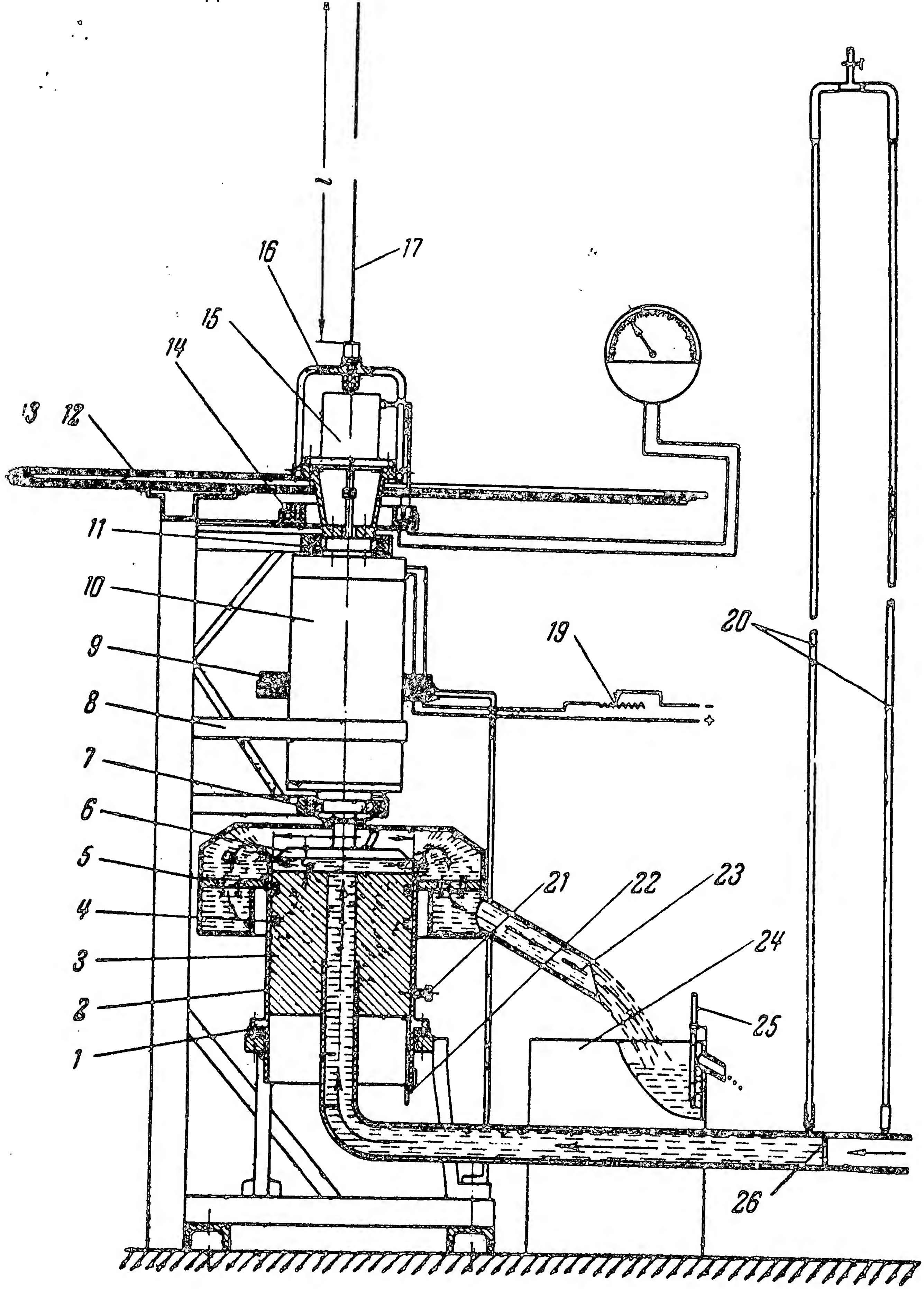

Fig. 3 - Sketch of experimental setup

(See Page 14, following, for names of numbered parts.)

NAMES OF NUMBERED PARTS, FIG. 3

1. Gasket (Spacer)
2. Casing
3. Piston (Plunger)
4. Packing ring
5. Box (Casing)
6. Disc
7. Radial thrust bearing
8. Brake band
9. Plexiglas ring with mercury filling
10. Electric motor
11. Thrust bearing
12. Indicator needle
13. Graduated dial
14. Plexiglas ring with mercury filling
15. Variable current generator
16. Motor suspension bracket
17. Dynamometer torsion spring
18. Voltmeter
19. Regulating rheostat
20. Orifice meter manometer piping
21. Gap width setting
22. Vernier (Nonius)
23. Valve (Stop)
24. Weigh tank
25. Mercury thermometer
26. Orifice

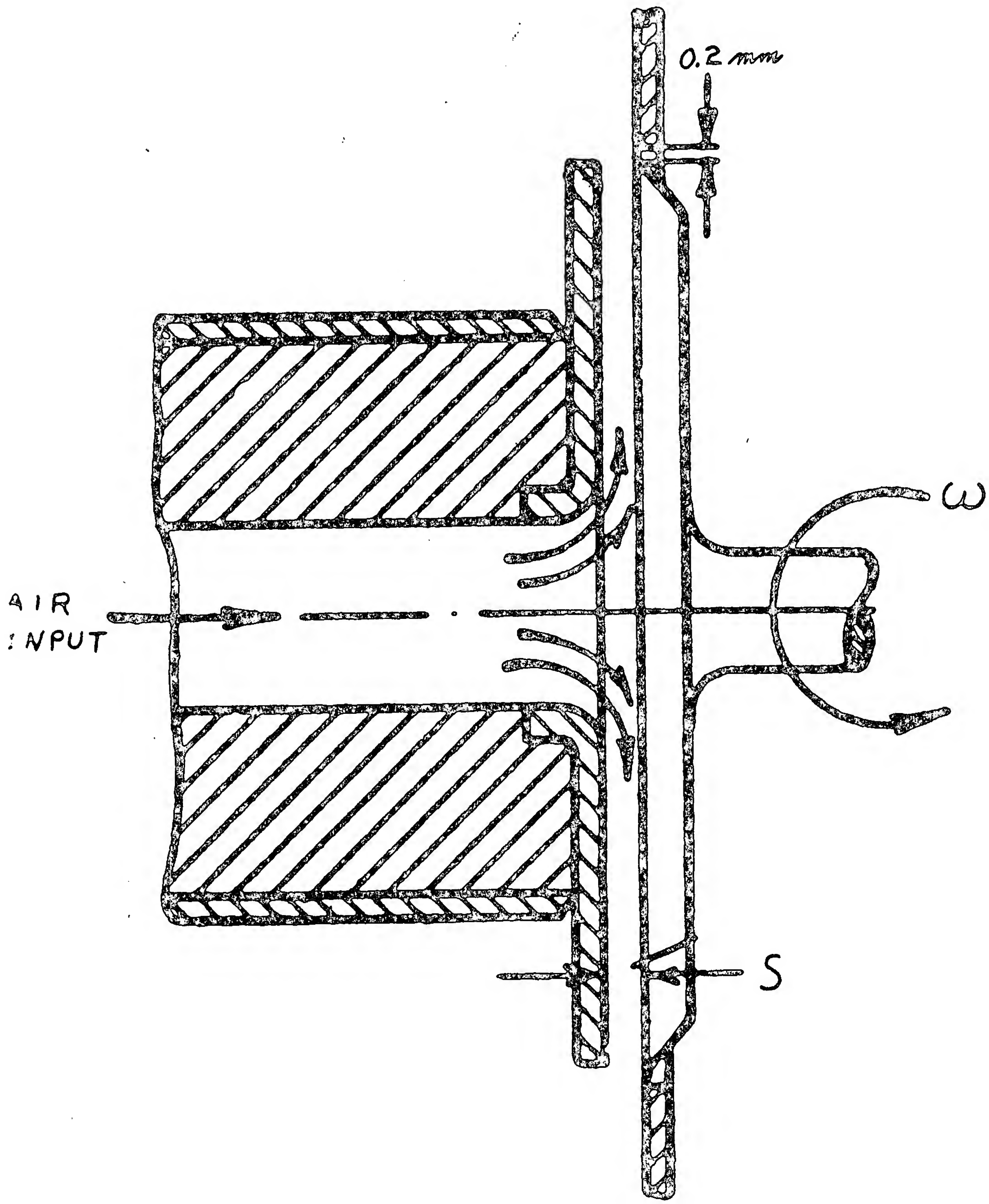

Fig. 4 - Scheme for introducing air to the disc

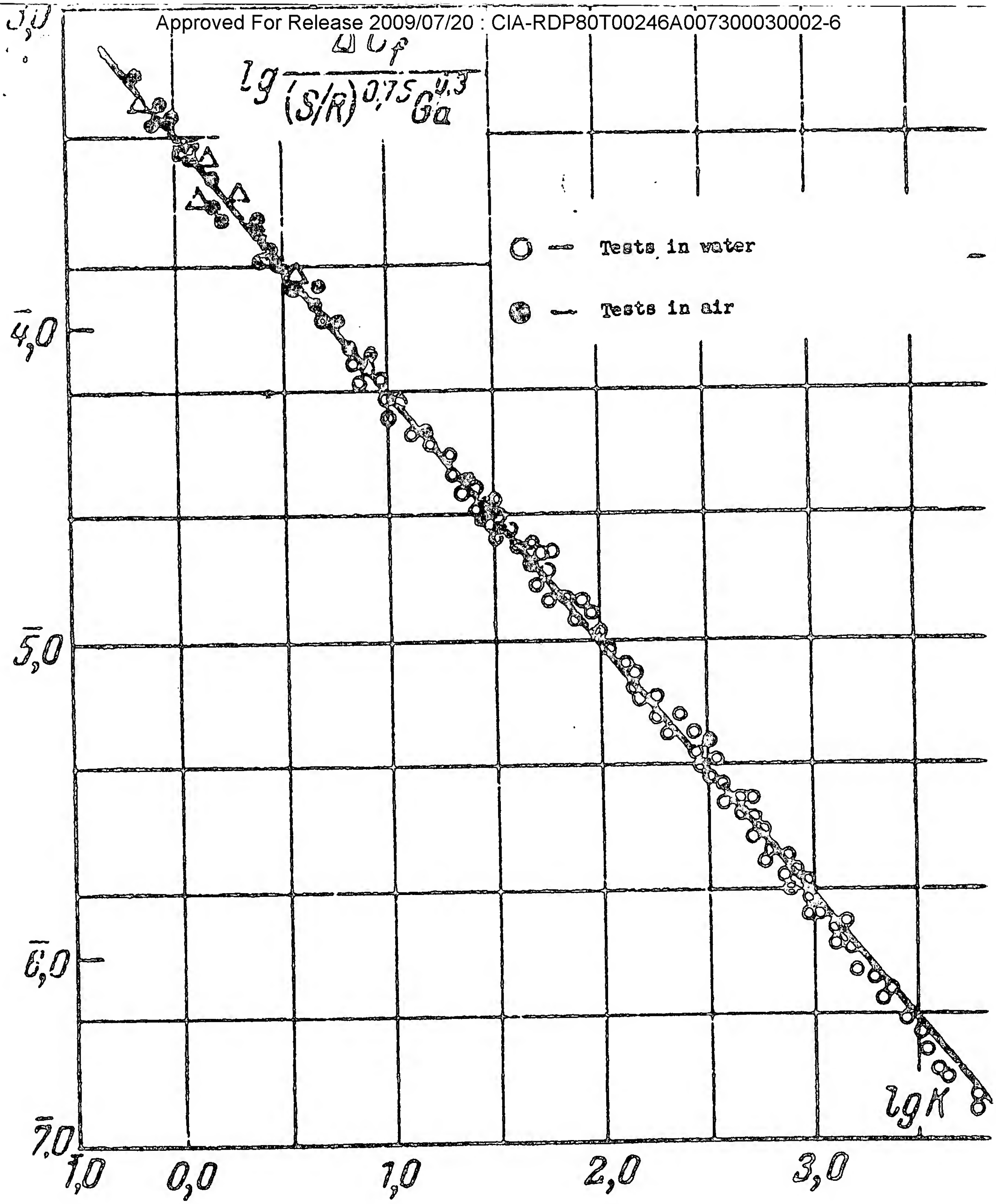

Fig. 5 - Relationship between $\lg \dfrac{\Delta C_f}{\left(\dfrac{S}{R}\right)^{0,75} Ga^{0.3}}$ от lg